A model for the analysis of pump start-up transients in Tehran Research Reactor

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Abstract

During the starting up period of a centrifugal pump the rotational speed is accelerated rapidly from standstill to its final speed and then established at the rated speed. While flow rate and total dynamic pressure rise are boosted accordingly. In the point of view of a theoretical treatment, pump start-up offers difficulty because both rotational speed and flow rate are variable. A far more difficulty is the linearization of the pump speed which is fundamentally impossible. Since a change in speed of the pump is one of transients cause leading to variation in the pressure and discharge. In order to predict the behaviour of the pump during start-up transient condition a mathematical model is developed. The model is based on a working practical approximation in that the total start-up time is reduced by one third. The influence of the two most important parameters, kinetic energy in the piping system and kinetic energy of the pump, is taken into account in the form of a ratio called hereafter an effective energy ratio. Preliminary test of the model was made on the existing piping systems related to nuclear reactors. As a case of study the safety of the Tehran Research Reactor (TRR) primary piping system, an open pool MTR-type research reactor, is considered. It is demonstrated that, by comparing the curves resulted from the numerical solution of the model in which design data of the TRR are used with the experimental characteristics curves of the TRR, the possibility of primary coolant pump turbining is exceptional.

Keywords: Nuclear research reactor; Pump start-up transient

1. Introduction

Accurate evaluation of rapid flow transients is an important factor in nuclear reactor design (Arker, 1956). Methods for the analysis of such transients may also find applications in other fields. Various assumed situations should be investigated which include the transients caused by fast opening and closing of valves and power failure to the centrifugal pump motors. During the start-up of a centrifugal pump and prior to the time in which steady-state flow is reached, certain transient conditions can produce heads and consequently require torques much higher than design. In some cases, selection of the motor and the pump must be based on start-up rather than on steady-state flow conditions. However, for an MTR pool type research reactors such as Tehran Research Reactor (TRR) it is necessary for safety purposes to thoroughly investigate normal start-up transients. For reactors with negative temperature
coefficients, a sudden increase in flow will result in a surge of reactor power generation which must be evaluated. Most of the investigations on flow transients are devoted to flow coastdown (Hamidouche et al., 2004; Housiadas, 2000) whilst very limited works have been done towards further understanding of pump start-up transients.

Complete characteristics for the prediction of behaviour of the pump during operating transients are discussed by Knapp and Pasadena (1937). The assumptions involved are investigated and experimental checks of their validity are offered. The analytical interrelationships between the hydraulic characteristics of the pumps and the pipeline are indicated. They did not consider water hammer problems during their studies, however.

Rapid flow transients in multiloops are studied by Arker and Lewis (1956). They proposed a general analytical technique, based upon the conservation of energy, which has also been applied to other situations such as the starting of pumps, the opening and closing of valves. They concluded that their method fulfils the need for highly accurate prediction of flow transients if sufficient steady-state information is available for all components, especially the pumps, and if water hammer effects are negligible.

Differential equations of motion for the rotating parts of the pump as well as the fluid in the pipes are derived by Kittredge and Princeton (1956). Methods of integrating the equations for the case of an assumed rigid fluid column and for the elastic fluid column are discussed. The methods of analytical solution have been applied to the power failure transient but they may be extended to other problems of abnormal operation.

Boyd et al. (1961) have written a mathematical simulation whereby system coolant flow and pump speed can be determined for various transient conditions for multiloop nuclear reactor system. They studied transients due to power

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**Nomenclature**

*Symbol quantity*

- \( A \) fluid flow cross-section (\( \text{m}^2 \))
- \( D \) pipe diameter
- \( D_e \) pipe diameter or equivalent channel hydraulic diameter (\( \text{m} \))
- \( f \) friction factor
- \( g \) gravity acceleration (\( \text{m/s}^2 \))
- \( \Delta p \) pressure difference
- \( h \) ratio of pump head to steady-state pump head (\( h = h_p/h_p(\text{ss}) \))
- \( h_p \) pump head (\( \text{m} \))
- \( I_p \) pump moment of inertia (\( \text{kg m}^2 \))
- \( k \) a constant defined in Eq. (8)
- \( m \) ratio of pump torque to steady-state pump torque
- \( M \) pump impeller torque (\( \text{N m} \))
- \( n \) pump rotational speed (\( \text{rpm} \))
- \( N \) rotation as fraction of rated rotation (\( n/n_{\text{ss}} \))
- \( q \) volumetric flow (\( \text{m}^3/\text{s} \))
- \( Q \) head as fraction of rated head (\( q/q_{\text{ss}} \))
- \( t \) time (\( \text{s} \))
- \( T \) normalized time (\( t/t_{2/3} \))
- \( v \) velocity (\( \text{m/s} \))
- \( V \) volume (\( \text{m}^3 \))
- \( \rho \) density (\( \text{kg/m}^3 \))

*Subscript*

- \( \text{ss} \) steady-state
- \( i \) referring to a portion of loop of uniform cross-section
- \( \text{p} \) pump
- \( \text{e} \) equivalent
- \( \text{f} \) fluid
failure, starting pumps in idle loops, and the opening of an active pump’s discharge valve. They emphasized all the components which are subject to the full primary coolant flow that offer resistance change in the system must be considered. Also, electrical torque as a function of speed is required for all analyses of start-up and coastdown transients. The kinetic energy of the fluid during start-up was taken into account in their studies. They have not compared their analytical results with experiments.

Flow coastdown in centrifugal pump systems was studied by Yokomura (1969). His method is applicable to a system with one reactor and any number of centrifugal pumps. The kinetic energy of rotating parts of pumps and motors was considered, but the electrical energy of the motors was out of consideration because power failure was assumed.

Start-up pressures in short pump discharge lines are investigated by Joseph and Hamill (1972). In particular, they concentrated on that type of pumping system which has no control valve but simply a check valve located immediately downstream from the pumps. The authors, in their analysis, neglect the kinetic energy given to the fluid during pump start-up.

In other study, an analysis of pump start-up transients has been accomplished by Grover and Koranne (1981). They developed a computer programme to solve an equation representing various torques during start-up numerically. They used characteristics of centrifugal pumps and electrical torque was neglected. Predictions of the programme have been verified experimentally, however.

In the current framework a mathematical model for analysing pump start-up transients is presented. In this model, differential equations for the fluid flow in the piping system and the motion of the rotating parts of the centrifugal pump are derived. Based on the assumption that the pump speed rises from rest to rated speed linearly, the differential equations are normalized. This is done by using time of fluid flow in pipes and start-up time of pump. The ratio of two-third start-up time of fluid flow in pipes to the start-up time of pump determines the transients during start-up.

Preliminary tests of the developed model were made on the existing published works on flow transients related to centrifugal pump start-up. Then, as a case study, Tehran Research Reactor is considered for the safety analysis. Prediction of the model has been experimentally verified by comparing the results obtained from the model with the TRR pump characteristics’ curves.

2. Centrifugal pump start-up transients

Many of the important transient analysis situations arising from hydraulic design are involved in the start-up of pumps and the opening and closing of valves associated with them. Apart from pipe rupture, deviation from steady-state flow in a piping system occurs because of a change in boundary conditions. There are many kinds of boundary conditions that introduce transients. For an MTR pool type research reactor piping system there are two boundary conditions, among the others, which are of prime importance. On priority these are pump start-up and changes in valve settings either accidental or planned. For pumps, the boundary condition takes the rotary inertia of moving parts into consideration.

With regard to pump characteristic curves, assuming the homologous relationships are holding true, a family of head-discharge curves may be drawn using rated speed as a parameter (Joseph and Hamill, 1972). The quicker pump start-up period will result in lower pipeline and pump discharges at the time when the pump attains rated speed (actual operating system). The pump characteristic curves, head-discharge with rated speed as a parameter, show that the pump will develop high pressures for these conditions. On the contrary, slow start-up period will approach a condition wherein rated flow and rated speed will be attained at the same time. However, in this case the pump will develop no pressure in excess of its rated speed.

Centrifugal pumps are commonly driven by amortisseur (squirrel cage) winding induction motors. Synchronous motors start as an induction motor utilizing the amortisseur winding with short-circuited rotor bars. These induction motors are frequently started without using reduced voltage (technically known as started across-the-line). This type of start produces a large starting torque from the induction motor. This in turn brings the pump up to speed quite rapidly. Induction motor torque characteristics are reviewed in more details (Skilling, 1962).

If the pump primarily has to develop pressure head for lifting up the liquid to a reservoir at a higher elevation, then pump start-up rate is dependent upon the flow in the pipeline. This effect may be set aside by considering a conservative estimate of effective start-up time. Generally, in most cases of centrifugal pumps the torque required at zero flow is less than or close to that which is required at rated flow. Consequently, employing a pump torque curve based on the assumption of shut-off conditions results in the calculation of a rapid acceleration. This method of computation is
conservative. In some cases of axial-flow pumps as well as mixed-flow pumps, however, the torque at zero flow is quite high as compared to the rated flow. For the aforementioned situations, the calculated start-up period may be non-conservative. Therefore, the computed start-up period must be adjusted downward to take into account for the smaller torques at finite flows.

In any pipeline where a pump has to circulate liquid into a reservoir with static head, the effective start-up period is actually less than that which is calculated. This is due to a seated check valve downstream of the pump which does not open until the pump develops a pressure head higher than the level of static head of the reservoir. Should the static head in reservoir be large, then the time necessary for the pump to attain a speed where it will develop static head will amount to a significant part of the total start-up period. As a result, the effective start-up period for the pressure rise calculation is the total start-up duration minus the time for the pump to attain static head of reservoir.

Results of a transient analysis simulating sudden power interruption to the pump motors have indicated that no pressures would occur that would exceed the initial steady-state condition (Joseph and Hamill, 1972). The authors in their analysis concluded that the maximum pressures are those which happen during pump start-up. Maximum design pressures for a pump discharge pipeline, where all the valves are open but the check valve, may be governed by water hammer effects caused by pump start-up. The maximum pressures may be reduced further by applying a method of slowing the pump induction motor acceleration. This can be done by increasing the total moment of inertia of the rotating parts either within the electrical limitations of the induction motor or by various electrical methods such as low voltage (part-winding) starts.

3. The mathematical description

A feasible and rigorous problem that arises in reactor safety analysis is the prediction of the system during transient pump start-up. During the start-up of a centrifugal pump and prior to steady-state flow being established, transient conditions control the system. These transient conditions often caused the pump to develop pressure head much higher than design. The system to be analysed, primary cooling system of the TRR, is shown in Fig. 1. The main technical specifications of the core are shown in Table 1. It comprised reactor core, hold-up tank, centrifugal pump, heat exchanger, and connecting pipes. To describe the pump transient a simplified approach, which renders the problem analytically solvable, is used in the present work. Nevertheless, it contained all the features of centrifugal pump characteristics.

The principle, upon which most of transients depend, is that the instantaneous performance of the pump for any given set of momentary conditions occurring during a transient is identical with the steady-state performance for those

![Fig. 1. Primary cooling system of TRR.](image)
same operating conditions. In this assumption two others are implied. First, two or more types of flow cannot exit within the pump for one given set of operating conditions, even momentarily. Second, the momentary accelerating forces exerted on the fluid within the pump during the transients are small in comparison to the forces required for normal steady operation at that particular state (Knapp and Pasadena, 1937). The interaction of the pump and the fluid in the connecting pipelines can be handled by a boundary condition. The homologous conditions, when the transient behaviour of a given pump is investigated, may be restated as follows. The ratio of retarding torque of the rotating parts to the square of pump speed and the ratio of pressure head developed by the pump to the square of pump speed are constant (Wylie and Streeter, 1993).

In the event of pump start-up, the flow behaviour is dominated by the inertia of the rotating parts and inertia of the fluid. The equation for pump dominated transients can be written as (Todreas and Kazimi, 1990):

\[
\frac{1}{g} \left( \sum \frac{L_i}{A_i} \right) \frac{dq}{dt} = h_p - \frac{1}{g} \left( \sum \frac{L_e}{D_eA_e^2} \right) \left( \frac{f}{2} \right) q^2
\]  

(1)

This equation indicates that rate of change of the fluid flow is controlled by the resistive factors of the loop and the pressure head developed by the pump. The effect of different inlet and outlet areas is included, within the first bracket.
on the right hand side, by an equivalent section. Head losses due to change in pipe diameter, pipe bends, and valves, may also be written similarly and are included in the same bracket. Time-dependent variables can be neglected from Eq. (1) during steady-state operation. In other words:

\[ \frac{1}{g} \left( \sum L_c \right) \left( \frac{f}{2} \right) = \frac{h_{p(\text{ss})}}{q^2_{(\text{ss})}} \]  

(2)

Centrifugal pump characteristic curves indicate that the retarding torque of the rotating parts and the pressure head developed by the pump, apart from being dependent upon square of pump speed, are also functions of the ratio of pump speed to velocity of flow. This assumption is confirmed to be true by experiments (see Fig. 1 of Takada et al., 1969). This implied that there is no slip between the pump impeller and fluid. In other words, the flow rate is proportional to the impeller angular velocity. Therefore, Eq. (1) can be rewritten as:

\[ \frac{1}{g} \left( \sum L_c \right) \frac{dq}{dt} = h_{p(\text{ss})} - \frac{q^2}{q^2_{(\text{ss})}} h_{p(\text{ss})} \]  

(3)

The method used in the present work for analysing pump start-up transients begins with prediction of coolant flow as a function of time. Upon failure of the power supply, the kinetic energy stored in the coolant and pump will be dissipated by frictional losses in the fluid system and frictional and electrical losses in the pump, respectively. The general technique will then be applied to pump start-up transients subsequently.

If we consider the case of pump failure or inadvertent pump stop during the reactor operation, the impeller does not influence the coolant flow. This in turn results in:

\[ \frac{1}{g} \left( \sum L_c / A_l \right) \frac{dq}{dt} = -\frac{q^2}{q^2_{(\text{ss})}} h_{p(\text{ss})} \]  

(4)

Solution of which is represented by Eq. (5).

\[ t = q^2_{(ss)} \frac{\sum L_c / A_l}{gh_{p(\text{ss})}} \left[ (q)^{-1} - (q_{(ss)})^{-1} \right] \]  

(5)

For analysing a centrifugal pump start-up event, a suitable working approximation is to work with a time in that the coolant flow in a given piping system diminished by one third of steady-state value (Wylie and Streeter, 1993). Alternatively:

\[ t_{2/3} = \frac{q_{(ss)} \sum L_c / A_l}{2gh_{p(\text{ss})}} \]

The aforementioned general technique includes non-dimensional parameters such as coolant flow rate, impeller speed, pressure drops and torques.

With \( T = t/t_{2/3} \) and \( Q = q/q_{(ss)} \), Eq. (3) will be non-dimensional accordingly:

\[ \frac{dQ}{dT} + \frac{1}{2} Q^2 = \frac{1}{2} \frac{h_{p}}{gh_{p(\text{ss})}} \]  

(6)

In addition, it is assumed that the developed pressure head across the pump is proportional to the square of impeller speed. Consequently:

\[ \frac{dQ}{dT} + \frac{1}{2} Q^2 = \frac{1}{2} N^2 \]  

(7)

where \( N \) is the ratio of transient to steady-state pump impeller speed.

The magnitude of the applied torque, furnished by the electric motor, depends generally upon the motor characteristics and associated start-up network. To produce a more or less uniform torque during speed change, for
simplification of the analysis, it is assumed that the external start-up network controls both the magnetic field and armature current. The pump start-up equation, assuming a constant applied torque, is given by:

\[ I_p \left( \frac{dn}{dt} \right) + kn^2 = kn_{ss}^2 \tag{8} \]

where the R.H.S. of Eq. (8) is a steady torque applied to the pump. Should air be used as a working fluid, during the start-up of the pump, rather than water there would be no retarding impeller torque. For this situation, the original pump equation should be rewritten as:

\[ I_p \left( \frac{dn}{dt} \right) = kn_{ss}^2 \tag{9} \]

Resulting in, \( t = I_p n / kn_{ss}^2 \). With \( n = n_{ss} \), the required time for a given pump to start-up is obtained:

\[ \tau_{2/3} = \frac{I_p}{kn_{ss}} \]

Non-dimensional Eq. (8) leads to:

\[ \frac{dN}{dT} = (\varepsilon - \varepsilon N^2) \tag{10} \]

where \( \varepsilon = t_{2/3} / \tau_{2/3} \). This ratio, \( \varepsilon \), has another physical meaning. As already mentioned, the boundary condition takes into consideration the rotary inertia of moving parts. The two most important variables influencing start-up transients are inertia of the rotating parts and inertia of the coolant fluid. These parameters are related to pump kinetic energy and kinetic energy of the fluid. The kinetic energy stored in the rotating parts during steady-state operation is:

\[ KE_p = \frac{1}{2} I_p n_{ss}^2 \]

Also, the kinetic energy stored in the flowing coolant fluid:

\[ KE_f = \frac{1}{2} \sum \frac{L}{A} \left( \frac{\rho g}{g} \right) q_{ss}^2 \]

The ratio of stored energy of coolant fluid to the stored energy in the pump is an effective energy ratio. Thus:

\[ \varepsilon = \frac{1}{2} \frac{KE_f}{KE_p} \eta_{ss}, \quad \text{where } \eta_{ss} = \frac{\rho gh_{p(\text{ss})} q_{ss}}{M_{pss} n_{ss}} \]

It is worthy to note that the time scale is non-dimensional with respect to the coolant fluid in the primary loop rather than the pump two-third time. The effect in pump energy, when the fluid parameters are fixed, is then properly shown.

The solution of Eq. (10) where \( N = 0 \) when \( t = 0 \) will be:

\[ N = \tanh \varepsilon T \]

By substituting in Eq. (7) and considering as initial condition \( Q = 0 \), the analytic solution that describes the pump start-up coolant flow is then derived through the following differential equation:

\[ \frac{dQ}{dT} + \frac{1}{2} Q^2 = \frac{1}{2} \tanh^2 \varepsilon T \tag{11} \]
4. Experimental and numerical validation

The Tehran Research Reactor primary cooling system is shown in Fig. 1. It consists of several sections representing the reactor core, heat exchanger, centrifugal pump, valves, and connecting piping. The analysis of pump start-up model is based on the simplifying assumption that the impeller torque and pump head are both proportional to the square of the pump speed. Pump characteristic curves indicate that in addition to the dependence upon the square of the pump speed the impeller torque and head are also functions of the ratio of velocity of flow to pump speed. The developed mathematical model is used to study the influence of moment of inertia of the rotating parts as well as fluid inertia on centrifugal pump start-up transient.

The numerical solution of Eq. (11) for different values of the effective energy ratio $\varepsilon$ is shown in Fig. 2. All of the variables shown in the figure are normalized. $Q$ is the ratio of the transient flow to steady-state flow, $T$ is the ratio of elapsed time to the loop two-third time, and $\varepsilon$ is the ratio of effective energy in the fluid to the effective energy stored in the pump. It is worthy to note that the time scale is normalized with respect to the coolant fluid in the primary loop rather than the pump two-third time. The effect in pump energy, when the fluid parameters are fixed, is then properly shown.

It is interesting to note the dependency of non-dimensional start-up time upon the effective energy ratio $\varepsilon$. The curves shown in Fig. 2 are drawn for a given primary piping system. When the energy stored in the rotating parts of the pump is very high, the effective energy ratio $\varepsilon$ will be very small; as shown in Fig. 2, the pump impeller will continue to pump the coolant fluid for a relatively long time. A relatively large value of an effective energy ratio $\varepsilon$ indicates a pump with a small amount of stored energy, however, or a pump which may not continue its pumping task for an extended time, but rather allows the coolant flow to decay rapidly under the influence of the friction forces in the primary cooling system. In this situation the impeller is susceptible to turbining, since a low-energy impeller is feature of a pump with small inertia.

Fig. 3 shows centrifugal pump start-up curves for different nuclear reactors’ primary piping systems, complementary data are shown in Table 2.

The figure shows that the transient situations change when the moment of inertia of the rotating parts of the pump is increased (for example see curves of Grover and Koranne, 1981). For a given $\sum L/A$ (a measure of inertia of the fluid contained in the piping) increasing moment of inertia results in decreasing the effective energy ratio $\varepsilon$. In other words, slow start-up times will approach a condition wherein rated flow and rated speed will be reached simultaneously. In this case, the pump will produce no pressure in excess of its rated head. Conversely, the faster start-up times will result
in lower pipeline and pump discharges at the time when the pump reaches rated speed. Also shown in the figure is the effect of increasing $P_L = A$ while keeping the moment of inertia fixed. From the figure it is clear that, for a given piping system, increasing $P_L = A$ will increase the effective energy ratio $\varepsilon$. It is also interesting to note that changes in $P_L = A$ do not have any appreciable influence on the rate of rise of speed, but it does influence the rate of rise of flow. Similar start-up flow pattern is shown by other curves.

The analytical start-up equation, Eq. (11), takes into account both the inertia of the rotating parts of the pump as well as the inertia of the coolant flow in the piping systems. However, the more exact solution of the problem, taking into account the pump characteristics, may be obtained from the following pump and flow equations:

$$\frac{dN}{dT} + m\varepsilon = \varepsilon,$$  \hspace{1cm} \text{where } N = 0, \text{ when } t = 0 \tag{12}

$$\frac{dQ}{dT} + \frac{1}{2}Q^2 = \frac{1}{2}h,$$  \hspace{1cm} \text{where } Q = 0, \text{ when } t = 0 \tag{13}

In Eq. (12), $m$ is the ratio of the transient torque to steady-state torque and in Eq. (13), $h$ is the ratio of transient pump head to steady-state pump head. The quantities $m$ and $h$ are the common pump operating characteristics and are often shown as a function of the ratio of discharge velocity to the pump speed.

### Table 2
Design data and effective energy ratio ($\varepsilon$) for different primary pipe flow loops

<table>
<thead>
<tr>
<th>Physical quantities</th>
<th>$N_R$ (rpm)</th>
<th>$H_R$ (m)</th>
<th>$Q_R$ (m$^3$/s)</th>
<th>$\eta$ (%)</th>
<th>$I_P$ (kg m$^2$)</th>
<th>$\sum \frac{1}{2}(m^{-1})$</th>
<th>$\varepsilon$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Arker and Lewis (1956)</td>
<td>3580</td>
<td>–</td>
<td>0.436</td>
<td>86</td>
<td>0.8</td>
<td>38.56</td>
<td>0.3425</td>
</tr>
<tr>
<td>Yokomura (1969)</td>
<td>1470</td>
<td>41.9</td>
<td>0.125</td>
<td>75.6</td>
<td>3.7</td>
<td>959</td>
<td>0.1131</td>
</tr>
<tr>
<td>Grover and Koranne (1981)</td>
<td>–</td>
<td>57.79</td>
<td>0.015729</td>
<td>83</td>
<td>0.2$^a$</td>
<td>6916$^d$</td>
<td>0.3250$^{(e)}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.5$^b$</td>
<td>13832$^e$</td>
<td>0.1300$^{(e)}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.0$^c$</td>
<td>27664$^f$</td>
<td>0.0650$^{(e)}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.0161$^{(a)}$</td>
<td>0.3250$^{(a)}$</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.6506$^{(a)}$</td>
<td></td>
</tr>
<tr>
<td>Tehran Research Reactor (1986)</td>
<td>1465</td>
<td>30.4803</td>
<td>0.138798</td>
<td>83</td>
<td>0.2</td>
<td>1536</td>
<td>0.9470</td>
</tr>
</tbody>
</table>
In the mathematical model, the assumption is made that the pumping head is always positive and that it approaches 0, as \( \varepsilon \) becomes large. The results obtained from present model which does not require detailed pump characteristics are compared with solutions that are obtained by using pump characteristics’ curves. The TRR primary cooling system pump is used for the purpose of comparison. Among four quantities involved in the characteristics, two of them are considered to be independent. For a given discharge \((Q)\) and speed \((N)\) including sign, \(h\) and \(m\) are determined from the pump characteristics (Wylie and Streeter, 1993). Also, two basic assumptions are made. (a) The steady-state characteristics hold for unsteady-state situations. Even though discharge and speed are changing with time, their values at an instant determine \(h\) and \(m\). (b) Homologous relationships are valid.

Two polynomials are needed to express \(h\) and \(m\) both of which are proportional to the square of pump speed. The existing TRR characteristics’ curve is for one rated speed. In many design situations the complete pump characteristics for different speeds are not available from the manufacturer, so one must obtain the characteristics’ curve from the available test data (Wylie and Streeter, 1993).

Based on the TRR characteristics’ curve corrections are made to diameter and speed of the other pumps (see Table 2 for design data). Fig. 4 shows head-flow characteristics’ curve for the TRR and three other homologous units. Then two non-dimensional coefficients are defined. The non-dimensional flow coefficient \(C_q\) which is defined as \(C_q = q/(\pi n D/60)D^2\) and the non-dimensional head coefficient or pressure developed by the pump is expressed as \(C_h = g \Delta h/(\pi n D/60)^2\), respectively.

Fig. 5 shows the steady-state hydrodynamic performance of pump impeller expressed as \(C_h\) vs. \(C_q\). Data are presented for a range of pump impeller speeds and a curve fit of all data is plotted as the solid line. The collapse of data onto a single curve for different pump impeller speeds and the small amount of scatter provide confidence in applying all the curves of Fig. 4 as the TRR characteristics’ curves for different pump impeller speeds. It also indicates that speed scaling effects are negligible.

Based on rated speed of the pump two polynomials representing the ratio of the heads and torques (homologous relations) are derived from the TRR pump characteristics’ curves and are represented as Eqs. (14) and (15), respectively.

\[
\frac{h}{N^2} = 1.275 + 0.308 \left( \frac{Q}{N} \right) - 0.983 \left( \frac{Q}{N} \right)^2 + 0.667 \left( \frac{Q}{N} \right)^3
\]

\[
\frac{m}{N^2} = 0.325 + 0.572 \left( \frac{Q}{N} \right) + 0.610 \left( \frac{Q}{N} \right)^2 - 0.347 \left( \frac{Q}{N} \right)^3
\]

![Figure 4](image_url)  
Fig. 4. Pressure vs. flow rate for a series of homologous pumps.
Using a fourth-order Runge-Kutta method, Eqs. (14) and (15) are solved numerically. The curves obtained from the model and from the TRR characteristic curves show similar flow pattern start-up. This is shown in Fig. 6. For a small value of the effective energy ratio $\epsilon$ the trends are expected to be very close. One would anticipate this, since low values of the effective energy ratio $\epsilon$ indicate high inertia impellers which require a longer time to be brought up to steady-state operating speed. During this period, the coolant fluid is brought up slowly to steady-state flow, with the result that there is no appreciable change in the ratio of speed of pump to velocity of flow. The constant-characteristic-based analytical solution assumes that this ratio remains constant. Consequently, for small values of $\epsilon$ the condition is closely satisfied.

However, pumps possessing large values of the effective energy ratio $\epsilon$ indicate the presence of impellers with low inertia. Therefore, during constant torque start-up, such impellers would approach steady-state operating speed much more rapidly than the coolant flow would approach steady-state flow.

Fig. 6 shows that at a higher value of the effective energy ratio $\epsilon$ the pump impeller speed goes beyond the rated speed after the start-up and then settles down to the rated speed. It may be noted that the amount by which pump impeller speed overshoots increases as $\sum L/A$ is increasing. However, pump impeller speed cannot go much beyond the...
rated speed. This is due to the fact that in an induction motor maximum pump, impeller speed is limited by synchronous speed.

It is interesting to note that the rotational speed of TRR primary cooling system pump is accelerated rapidly from zero speed to its maximum speed and then established at the steady-state speed. As one would expect the flow rate follow the same situation accordingly. Similar behaviour is shown in the work of Grover and Koranne (1981: Fig. 3)

5. Conclusions

Accurate assessment of rapid flow transients during primary cooling system pump start-up is an important topic for both design and safety of nuclear reactors. During the starting of the centrifugal pump certain transient conditions produce head which necessitate torque much higher than design. Since the primary cooling piping system possesses an appreciable amount of coolant fluid the inertia of the coolant fluid mass could produce a significant resistance towards any change in accelerating the flow. The piping system having a pump with a high moment of inertia resulted in an extended time for the coolant to brought up slowly to steady-state flow. With the pump having a small impeller moment of inertia, during start-up, such a pump impeller will approach steady-state operating speed much more rapidly than the coolant fluid would approach steady-state flow. In the current framework, a mathematical model was developed in the present work to predict pump start-up transients. Preliminary calculations have been performed for the existing nuclear research reactor piping system and the results show fair agreement between the prediction of the model and the experimental pump characteristic curves. Comparison of the results obtained from the model with those obtained from the TRR experimental characteristic curves. The curves show similar flow start-up behaviour.

The present work highlights the importance of pump start-up transients in the primary cooling system of the TRR and similar systems. A further study is needed to consider the problem even deeper by including the effect of rapid closure of a check valve and possible water hammer occurrence. Perhaps the latter study would leads to a more realistic representation of pump start-up transients.

References